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# Transmission Efficiency Measurements and Correlations With Physical Characteristics of the Lubricant

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TRANSMISSION EFFICIENCY MEASUREMENTS AND CORRELATIONS  
WITH PHYSICAL CHARACTERISTICS OF THE LUBRICANT

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**SUMMARY**

Data from helicopter transmission efficiency tests have been compared to physical properties of the eleven lubricants used in those tests. The tests were conducted with the OH-58 helicopter main rotor transmission. Efficiencies ranged from 98.3 to 98.8 percent. The data was examined for correlation of physical properties with efficiency. There was a reasonable correlation of efficiency with absolute viscosity if the viscosity was first corrected for temperature and pressure in the lubricated contact. Between lubricants, efficiency did not correlate well with viscosity at atmospheric pressure. Between lubricants, efficiency did not correlate well with calculated lubricant film forming capacity. Bench type sliding friction and wear measurements could not be correlated to transmission efficiency and component wear.

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**INTRODUCTION**

In refs. 1 and 2 results were presented from efficiency testing of eleven different lubricants in an OH-58 main rotor transmission. The tests showed that the efficiency ranged from 98.3 to 98.8 percent, depending on the lubricant used. Furthermore, the efficiency for a given lubricant showed an increase with increasing inlet temperature with two exceptions, where the efficiency decreased with increased temperature. Since temperature affects viscosity, there was an increase in efficiency with a decrease in viscosity for a given lubricant, but no correlation among all the lubricants as a group was found.

The generally high efficiency was no surprise since it has long been recognized that the mechanical efficiency of helicopter power trains is quite high. Usually a planetary reduction has 3/4 percent loss, and a single bevel or spur gear mesh has 1/2 percent loss (ref. 3). Compared with the large amounts of power available from the engines of a helicopter, it may seem that fractions of a percent of the power lost in the power train path are inconsequential. However, the impact of higher losses is to require larger and heavier oil cooling systems. This effect adds to reduce helicopter payload and reduce survivability in a hostile area. Moreover, when the results from the work presented in refs. 1 and 2 are considered where there was as much as 50 percent variation in power losses among the lubricants tested, the impact on oil cooler weight, size, and vulnerability become very significant. By proper selection of lubricant, the operating envelope and payload capacity of the helicopter can be improved.

There are many factors which act together in causing the power loss in a helicopter transmission which is a rather complicated assembly of gears, shafts, seals, and bearings. In a typical application it is expected that sliding, windage, churning and pumping losses all play a role, as do a variety of physical parameters of the lubricant which are important in the particular mechanism of lubrication.

There have been significant contributions to the theory of power losses in transmission components. Martin (refs. 4 and 5) considered power loss between gear teeth. Townsend, Allen and Zaretsky (ref. 6) considered bearing power loss. Anderson and Loewenthal (ref. 7) gave a comprehensive analytical treatment of power loss in gear sets, including gear losses and bearing losses. Townsend and Akin (refs. 8 to 10) have concluded that efficiency and cooling in gear sets is optimum with radially directed lubricant jets on the exit side of the gear mesh. Murphy et al., (ref. 11) in their study of low-speed worm drives have concluded that synthetic oils with the lowest traction coefficients give the highest efficiencies.

The effort described in refs. 1 and 2 has been continued by investigating the physical properties of the lubricants. Results of the properties and characterization of the lubricants, including pressure-viscosity and friction effects, are reported in ref. 12.

In view of the aforementioned progress, the objective of the work presented herein was to summarize and compare the previously measured helicopter transmission efficiencies with the newly available physical characteristics of the lubricants.

Specifically, the pressure-viscosity effects and the friction effects were examined for possible correlation with measured transmission efficiency. The results of that investigation are reported herein.

#### APPARATUS, SPECIMENS, AND PROCEDURE

##### Transmission Test Stand

Figure 1 shows the NASA 500 hp helicopter transmission test stand, which was used to run the efficiency tests. The test stand operates on the "four-square" or torque regenerative principle, where mechanical power is recirculated around the closed loop of gears and shafting, passing through the test transmission. A 149 kW (200 hp) SCR controlled dc motor is used to power the test stand and control the speed. Since the torque and power are recirculated around the loop, only the losses due to friction have to be replenished.

An 11 kW (15 hp) SCR controlled dc motor driving against a magnetic particle clutch is used to set the torque in the test stand. The output of the clutch does not turn continuously, but only exerts a torque through the speed reducer gearbox and chain drive to the large sprocket on the differential gear unit. The large sprocket is the first input to the differential. The second input is from the upper shaft which passes concentrically through the hollow upper gear shaft in the closing end gearbox. The output shaft from the differential gear unit is the previously mentioned hollow upper gear shaft of the closing end gearbox. The torque in the loop is adjusted by changing the electrical field strength at the magnetic particle clutch. The 11 kW (15 hp) motor was set to turn continuously at 70 rpm.

The input and output shafts to the test transmission are equipped with speed sensors, torque meters, and slip rings.

Figure 2 is a schematic of the efficiency measurement system. The system allows the helicopter transmission to be operated in a thermally insulated environment with provisions to collect and measure that heat generation due to mechanical power losses in the transmission. In this schematic, the instrumentation used to measure torque and speed, and hence power input to the test transmission is not shown. The original oil-to-air heat exchanger which is standard flight hardware was replaced with an oil-to-water heat exchanger so as to allow more precise measurements of the heat rejection during an efficiency test run. By using the water to remove heat, any uncertainty of the correct value for specific heat of the oil was removed.

Figure 3 shows the test transmission mounted in the test stand. Figure 4 shows the test stand with the insulated housing around the test transmission. Thermocouples were placed at various locations inside the insulated housing to verify the adequacy of the insulation.

##### Test Lubricants

All the lubricants were near to the 5 to 7 centistoke range in viscosity and were qualified for use or considered likely candidates for use in helicopter transmissions. All the lubricants were tested (ref. 12) using new and used samples after completion of all efficiency test runs. Tables I to X, list the test lubricants, their specification, basestock characterization, physical properties, and chemical analysis performed. The methodology used is available in ref. 12.

##### Test Transmission

The test transmission was the main rotor transmission from the U.S. Army's OH-58 light observation helicopter as described in ref. 13 and shown in Fig. 5. The transmission is rated for 210 kW (270 hp) continuous duty and 236 kW (317 horsepower) at takeoff for 5 min. The 100-percent input speed is 6060 rpm. The input shaft drives a 19 tooth spiral bevel pinion. The pinion meshes with a 71 tooth gear. The input pinion shaft is mounted on triplex ball bearings and one roller bearing. The 71 tooth bevel gear is carried on a shaft mounted in duplex ball bearings and one roller bearing. The bevel gear shaft drives a floating sun-gear which has 27 teeth. The power is taken out through the planet carrier. There are three planet gears of 35 teeth which are mounted on spherical roller bearings. The ring gear (99 teeth) is splined to the top case and therefore is stationary. The overall gear ratio is 17.44:1 reduction.

The planet bearing inner races and rollers are made of AISI M-50 steel. The outer races and planet gears, which are integral, are made of AISI 9310. The cage material is 2024-T4 aluminum. The gear shaft duplex bearing material is CVM 52CB. All other bearings are made of AISI 52100 with bronze cages. The sun gear and ring gear material is Nitralloy N (AMS6475). The input spiral bevel gear-set material is AISI 9310. Lubrication is supplied through jets located in the top case, with circulation provided by an integral pump.

### Test Procedure

Before the start of each efficiency test, the transmission and heat exchanger were cleaned out with solvent and the transmission components were visually inspected. Gear tooth surfaces were photographed. The transmission was then assembled and mounted in the test stand and filled with oil. The rig was run briefly to check for oil leaks. Then the loose fill insulation was added, filling the plexiglass box to completely surround and thermally insulate the test apparatus and transmission.

Efficiency test runs were made with the oil inlet temperature controlled to within less than one degree kelvin. Tests were run at oil inlet temperatures of approximately 355 K (180° F) and 372 K (210° F). The torque on the input shaft was 352 N-m (3118 lb-in) for each run. The input speed was 6060 rpm. This corresponds to the full power condition on the test transmission. The oil inlet and oil outlet temperatures were monitored until equilibrium conditions were established, which generally took about 20 to 30 min. The efficiency tests were then started. Water was collected in the weighing tank and data were recorded for total water weight, inlet and outlet temperatures for the water and oil, and flow rate for the water and oil. Vibration spectrum records were taken once each minute for a total test time of approximately 30 min for each test temperature.

After the tests were completed the transmission was disassembled, cleaned and visually inspected for changes in the gear and bearing surfaces. Photographic records were made. The lubricant was saved for later analysis. The efficiency was calculated from the heat balance on the water that flowed through the heat exchanger.

### RESULTS AND DISCUSSION

The experimentally determined efficiencies are listed in Table XI and plotted against oil inlet temperature in Fig. 6. The range of efficiencies varied from 98.3 to 98.8 percent. This is an overall variation in losses of almost 50 percent, relative to the losses associated with the maximum efficiency measured.

In general, the higher test temperature for a given lubricant yielded a higher efficiency. The exceptions were with lubricants E and C, which were different types of synthetic lubricant. Lubricant G, being more viscous than the other lubricants could not be tested at the targeted oil inlet temperature. This was because the heat generated could not be removed with the existing water/oil heat exchanger. The test temperature increased to 378.5 K with the heat exchanger at full water flow capacity. At the higher temperature the efficiency for oil G was consistent with the efficiencies for the lower viscosity oils. The two automatic transmission fluids (A and B) and the Type I Synthetic Gear Lubricant (E) yielded significantly lower efficiencies as a group.

For meshing gear teeth, pure rolling exists at the pitch point, with increasing amounts of relative sliding as the contact point moves away from the pitch point. (Based on the research presented in ref. 7, the major power loss in the transmission has been determined to be due to sliding in the gears and bearings.) For higher amounts of sliding in an elastohydrodynamic contact, Couette flow predominates. Since the power loss for Couette flow with a Newtonian fluid is proportional to the absolute viscosity, a possible correlation of efficiency and viscosity was investigated. In Fig. 7 the efficiencies are plotted against the lubricant absolute viscosity at the inlet temperature. The correlation of efficiency with viscosity is rather limited. It is interesting to note that while the Mil-L-7808 lubricant (lubricant H) was the lowest viscosity oil, the efficiency was no better than the Mil-L-23699 lubricants (lubricants C, D, I, and K). By the plotted results, it is clear that viscosity variation is not the primary reason for the varying efficiencies between the different lubricants. But there is a general trend to higher efficiency for lower viscosity for all the lubricants except C and E. The slope of the aforementioned trend is identical for a large number of the lubricants.

While it is true that the elastohydrodynamic lubrication film thickness is determined by the fluid properties at the inlet, the traction and friction properties are determined by the properties within the contact area itself. Therefore, it may be more correct to use viscosity that is obtained from a viscosity equation that takes into account the pressure effects by using the Barus equation (ref. 14). A representative pressure of 0.6 GPa was used to correct the viscosity via the Barus relation. The corrected viscosity was investigated to determine if a correlation with efficiency existed. The results are shown in Fig. 8. The correlation is better than that in Fig. 7 in that lubricant H is now included in the major trend of decreasing efficiency with increasing viscosity at contact pressure conditions. Lubricants C and E continue to have reverse trends from the majority of other lubricants. The data for lubricants F and G which are the only synthetic hydrocarbon type lubricants are above the major trend line that was fitted to the lubricants.

In hydrodynamic lubrication, Couette flow predominates, but in elastohydrodynamic lubrication, (for pure rolling) Poiseuille flow is predominant. For Poiseuille flow, the power loss is proportional to the lubricant film thickness. Therefore, a possible correlation of efficiency with lubricant film thickness forming capacity at the inlet temperature was investigated. The film forming capacity is the product of absolute

viscosity (at inlet conditions) and pressure viscosity exponent, each raised to the 0.68 and 0.49 power, respectively (ref. 15). Results that are very similar to those of Fig. 7 were obtained when efficiency was plotted against the elastohydrodynamic film forming capacity. It can be concluded that the correlation of efficiency with film forming capacity is also rather limited.

In order to relate transmission efficiency to some readily measurable physical property of the lubricants, the friction coefficient of the lubricants was measured using the LFW-1 tester (ref. 12). There was much scatter in the friction data from this test (table X). The mean value for friction was examined for any correlation with calculated lubricant film thickness or viscosity at 373° C and 0.55 GPa which were the conditions of the LFW-1 test. There was a general linear regression trend of increasing friction with increasing viscosity which agrees with pure hydrodynamic fluid flow. In addition, there was a general linear regression trend of increasing friction with increasing film thickness which agrees with the previously mentioned postulate of Poiseuille flow in conjunction with elastohydrodynamic lubrication. When the efficiency was plotted against the mean friction coefficient for 373 K (212° F) it was found that a linear regression trend of decreasing efficiency with increasing friction coefficient existed. However, there was much scatter in the data. Additionally, for an individual lubricant, efficiency could not be determined based on the LFW-1 type measurement of friction coefficient. Furthermore, severe wear was observed on the LFW-1 test specimens. Visual inspection of the transmission components after each test run showed no indications of wear or degradation (ref. 1). In fact, the black oxide coating which was placed on the gear surfaces during manufacturing was hardly worn off. Hence, it must be concluded that bench type sliding friction and wear measurements cannot be correlated to transmission efficiency and component wear.

It is believed that the lower efficiencies for lubricants A, B, and E as well as the mentioned reverse trends of lubricants C and E are related to traction coefficient characteristics which are a function of the lubricants non-Newtonian constitutive relations as affected by lubricant base stock and additive package chemistry. The lubricant constitutive relations were studied by Höglund (ref. 16) and Kuss, et al. (ref. 17). The lubricant rheology study of Höglund was aimed at defining the dependence of lubricant traction behavior as a function of pressure and temperature. Höglund's measurements (ref. 16) were made for a broad sample of lubricants which included many of the types used in this study. The constitutive relations were characterized as follows. There was generally a limiting shear stress, beyond which the fluid was unable to support a stress that was calculated by the Newtonian expression where stress is proportional to strain rate. There was also a critical pressure or "solidification" pressure beyond which the lubricant limiting shear stress increased linearly with pressure. The solidification pressure increased with temperature. In Höglund's study, the ranking of lubricants with increasing solidification pressure at 373 K (212° F) was a synthetic traction fluid (1.07 GPa), a lithium soap grease (1.37 GPa), three paraffinic mineral oils (1.48, 1.66, 1.77 GPa), and finally the synthetic hydrocarbon and synthetic ester lubricants which did not solidify up to the limit of the test rig (2.2 GPa) at the 373 K (212° F) temperature. These synthetics did solidify at lower temperatures which in one case was as low as 313 K (104° F) for a polyalphaolefin/polyol ester synthetic lubricant.

What is significant about Höglund's results in relation to the present study, is that he shows there can be a large difference in the manifested frictional losses among various lubricants at the same pressures and temperatures as a result of the solidification pressures being different. It is believed that the lower efficiency with lubricants A, B, and E are due to there being a lower solidification pressure for these lubricants.

In the study of Kuss, et al. (ref. 17) it was shown that the addition of 9.6 percent sulphur to a base stock caused a drastic change in the viscosity versus pressure characteristics. For the base stock, there was generally an exponential trend of increasing viscosity with pressure up to 200 MPa (which is as far as the data points were taken). The addition of sulphur produced a knee in the viscosity versus pressure relation beyond which the viscosity increased even more rapidly with pressure. For the range of temperatures 298 to 323 K (77° to 122° F) investigated, the knee in the curve ranged from 40 to 190 MPa (276 to 1310 psi). What is significant about this in relation to the efficiency measurements presented here is that the measured viscosities presented in Table III and used for the possible correlation in Fig. 7 are for atmospheric pressure only. The viscosity at high pressures such as exist in the gear and bearing contact regions would be different, from that calculated using the pressure-viscosity coefficient with the Barus equation. Hence, an improved correlation may be obtained if the lubricant rheological properties that affect traction measured under conditions of pressure and temperature representative of the efficiency test conditions reported herein.

The reason for the decrease in efficiency with increase in temperature for lubricants C and E is unknown, but may be related to increased activity of the particular additive packages at the higher temperatures which may cause rheological changes in the fluid in conjunction with formation of chemically absorbed surface films. Lubricant E (table IX) has large amounts of chlorine, zinc, sulphur and barium which are indicative of large amounts of antiwear and detergent additives being present.

Tables VI to X give the comparison between the lubricant analyses performed before and after the efficiency test runs. It is noticed that lubricants A and C showed significant increases in the iron content (table VI). Also, lubricant E showed a strong acid value before and after the test runs (table VII). These three lubricants were among the ones giving deviant performances for efficiency.

#### SUMMARY OF RESULTS

Data from helicopter transmission efficiency tests have been reviewed and compared with data characterizing the physical and chemical properties of the lubricants used in the transmission. The transmission efficiency tests were conducted using eleven different lubricants in the NASA Lewis Research Center's 500 hp torque regenerative helicopter transmission test stand. The test transmission was the OH-58A helicopter main transmission. The mechanical power input to the test transmission was 224 kW (300 hp) at 6060 rpm. Tests were run at oil-in temperatures of 355 K (180° F) and 372 K (210° F). The efficiency was calculated from a heat balance on the water running through an oil-to-water heat exchanger while the transmission was heavily insulated. The test lubricants were analyzed for their physical and chemical properties. Newly available data on pressure-viscosity characteristics as well as friction data from LFW-1 type testing were examined for possible correlation with the efficiency data.

The following results were obtained.

1. There was a reasonable correlation of efficiency with absolute viscosity (corrected for temperature and pressure in the contact).
2. Between lubricants, efficiency did not correlate well with absolute viscosity at atmospheric pressure.
3. Between lubricants, efficiency did not correlate well with calculated lubricant film forming capacity.
4. Bench type (LFW-1) sliding friction and wear measurements could not be correlated to transmission efficiency and component wear.

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TABLE I. - TEST LUBRICANT TYPES

Lubricant NASA code (AFLRL code)	Specification	Generic type/Basestock
A (11252)	DEXRON II GM 6137-M	Automatic transmission fluid/mineral oil
B (11268)	DEXRON II GM 6137-M	Automatic transmission fluid/mineral oil
C (11250)	MIL-L-23699	Turbine engine oil/ester (PE)
D (11254)	MIL-L-23699	Type II synthetic gas turbine engine oil/ester (PE)
E (11256)		Formulated gear lubricant/dibasic acid ester
F (11258)		NASA gear test lubricant - synthetic paraffinic with antiwear additives/synthetic hydrocarbon (PAO)
G (11260)	MIL-L-2104C MIL-L-46152	Synthetic fleet engine oil/mixture of 80 percent synthetic hydrocarbon (PAO) and 20 percent ester (TMP)
H (11262)	MIL-L-7808	Turbine engine oil/ester (TMP)
I (11264)	MIL-L-23699	Type II turbine engine oil/mixture of 50 percent TMP ester and 50 percent PE ester
J (11270)	MIL-L-23699	Type II turbine engine oil/ester (PE)
K (11266)		Turbine engine oil/mixture of 99 percent PE ester and 1 percent DPE ester

PE = pentaerythritol

TMP = trimethylolpropane

PAO = polyalphaolefin

DPE = dipentaerythritol

TABLE II. - SPECIFIC GRAVITY DATA ACCORDING TO  
 ANSI/ASTM SPECIFICATION D-1481, API  
 GRAVITY ACCORDING TO ANSI/ASTM  
<sup>a</sup>SPECIFICATION D-1298  
 (DATA FROM REF. 12)

Lubricant code	Specific gravity at listed temp			API gravity 288 K
	313 K	355 K	373 K	
A	0.8620	0.8558	0.8514	29.8
B	.8626	.8548	.8546	29.9
C	.9973	.9862	.9843	8.2
D	.9868	.9768	.9746	9.7
E	.9322	.9211	.9201	17.7
F	.8262	.8108	.8088	36.0
G	.8629	.8536	.8527	29.6
H	.9442	.9320	.9313	15.7
I	.9659	.9568	.9546	12.8
J	.9856	.9759	.9747	10.1
K	.9829	.9721	.9725	10.3

<sup>a</sup>ANSI/ASTM, American National Standards Institute/American Society for Testing and Materials

TABLE III. - KINEMATIC VISCOSITY DATA  
 ACCORDING TO ANSI/ASTM  
 SPECIFICATION D-455  
 (DATA FROM REF. 12)

Lubricant code	Viscosity at listed temp. cSt		
	313 K	355 K	373 K
A	37.48	10.48	7.01
B	33.15	9.64	6.52
C	26.40	7.69	5.13
D	26.17	7.50	5.00
E	33.91	8.91	5.87
F	28.01	8.15	5.36
G	56.65	15.05	9.83
H	13.16	4.73	3.38
I	24.19	7.18	4.85
J	24.76	7.23	4.89
K	26.39	7.61	5.09

TABLE IV. - SPECIFIC HEAT DATA DETERMINED BY  
DIFFERENTIAL SCANNING CALORIMETRY  
(DATA FROM REF. 12)

Lubricant code	Specific heat at listed temperature					
	313 K		373 K		413 K	
	Cp	$\sigma$	Cp	$\sigma$	Cp	$\sigma$
A	0.42	0.091	0.42	0.12	0.44	0.14
B	.50	.048	.50	.051	.49	.07
C	.33	.097	.32	.097	.32	.091
D	.33	.071	.34	.072	.34	a.084
E	.68	.11	.73	.13	.76	.20
F	.53	.12	.54	.13	.54	.14
G	.50	.091	.47	.058	.42	.059
H	.37	.036	.30	.037	.31	.094
I	.53	.060	.47	.039	.44	a.075
J	.47	.031	.48	.030	.49	.030
K	.44	.073	.38	.076	.34	.075

<sup>a</sup>For calculation of Cp and and  $\sigma$  (std. deviation) one value, inordinately different from the others, was discarded. Thus, four values rather than five were used to determine these data.

TABLE V. - PRESSURE-VISCOSITY COEFFICIENTS  
FOR TEST LUBRICANTS EXPRESSED AS  
RECIPROCAL ASYMTOTIC ISOVISCOSUS  
PRESSURE (DATA FROM REF. 12)

Lubricant code	Reciprocal asymptotic isoviscous pressure $\alpha^*$ , GPa <sup>-1</sup> at listed temperature		
	313 K	373 K	423 K
A	15.37	11.72	10.22
B	14.96	11.85	10.34
C	11.63	10.03	8.81
D	12.43	9.94	8.71
E	15.53	11.51	9.88
F	13.44	11.14	9.53
G	13.80	11.34	10.36
H	11.53	9.14	7.95
I	12.08	9.24	8.34
J	11.96	9.23	8.30
K	11.40	9.50	8.32

TABLE VI. - TOTAL IRON ANALYSIS BY  
CALORIMETRIC METHOD (REF. 18)  
(DATA FROM REF. 12)

Lubricant code	Iron content, ppm	
	New	Used
A	1	4
B	<1	<1
C	1	6
D	<1	1
E	<1	1
F	<1	2
G	2	3
H	<1	1
I	<1	<1
J	<1	<1
K	<1	<1

TABLE VII. - LUBRICANT ACID  
ANALYSIS ACCORDING TO  
ANSI/ASTM SPECIF-  
ICATION D-664  
(DATA FROM REF. 12)

Lubricant code	Total acid number Mg KOH/g	
	New	Used
A	0.54	0.54
B	.62	.62
C	.01	.02
D	.07	.07
E	<sup>a</sup> 15.8	<sup>a</sup> 15.7
F	.42	.51
G	3.2	3.5
H	.34	.34
I	.34	.38
J	.51	.38
K	.48	.43

<sup>a</sup>Strong acid value = 7.1  
on samples

TABLE VIII. - PARTICULATE CONTAMINATION COUNT ACCORDING  
TO SAE AEROSPACE RECOMMENDED PRACTICE  
ARP 598A (DATA FROM REF. 12)

Lubricant code	Number of particles/100 ml					
	Particle sizes in micrometers					
	5-15	15-25	25-50	50-100	100	Fibers
A New	17	2	2	4	10	12
Used	4	1	6	7	11	10
B New	6800	2980	200	40	44	112
Used	49	51	27	23	16	18
C New	72	36	18	12	10	7
Used	4	1	2	1	5	9
D New	685	275	35	22	15	20
Used	200	65	38	24	21	39
E New	120	60	23	25	22	33
Used	44	7	10	13	12	19
F New	60	16	30	13	7	22
Used	475	8	2	5	6	52
G New	49	39	45	38	34	78
Used	4740	10	11	9	6	34
H New	1780	72	45	40	25	32
Used	1850	118	108	60	52	62
I New	54	23	17	16	4	19
Used	840	660	450	210	80	120
J New	47	22	10	7	12	18
Used	36	18	14	8	11	29
K New	185	175	100	70	35	45
Used	105	48	35	21	20	22

TABLE IX. - WEAR METALS TEST RESULTS USING X-RAY FLUORESCENCE FILTER METHOD (REF. 19) (DATA FROM REF. 12)

Lubricant code	Elements (PPM)													Limit(3) of Detection (PPM)	
	Mg	Al	Cl	Fe	Ni	Cu	Pb	Zn(1)	P(2)	S(2)	Ca(2)	Ba(2)	Si	Mn	
A	New	0.48	-----	2.47	----	-----	0.21	-----	0.18	4.71	-----	0.23	-----	-----	0.11
	Used	-----	5.91	1.12	0.51	0.10	0.14	-----	0.11	.17	1.12	-----	0.12	-----	.09
B	New	0.86	-----	1.80	----	-----	-----	0.88	0.47	10.40	-----	-----	0.33	-----	0.21
	Used	.60	4.00	1.90	0.57	-----	-----	-----	.74	.27	7.40	-----	-----	.90	-----
C	New	0.28	-----	0.73	0.13	-----	-----	-----	0.26	-----	-----	-----	-----	-----	0.09
	Used	-----	2.97	1.04	2.19	0.21	0.12	-----	0.15	.19	0.20	-----	-----	-----	0.09
D	New	0.27	-----	0.90	----	-----	-----	-----	0.16	-----	-----	-----	-----	-----	0.11
	Used	-----	12.7	2.08	1.16	0.24	0.19	0.20	0.20	.71	0.51	-----	-----	-----	.15
E	New	0.16	0.19	7.57	0.10	-----	-----	1.28	7.27	2.15	13.01	0.29	10.16	-----	0.09
	Used	.12	1.69	1.61	.26	-----	0.11	-----	3.71	.94	4.29	-----	2.43	-----	.09
F	New	0.31	-----	0.45	----	-----	-----	-----	0.19	7.08	-----	-----	-----	-----	0.10
	Used	5.36	-----	2.49	----	-----	-----	-----	2.42	51.0	-----	-----	-----	-----	.55
G	New	1.31	-----	4.91	----	-----	-----	-----	1.51	0.70	5.29	8.69	-----	-----	0.43
	Used	.39	0.67	1.49	0.22	-----	-----	-----	0.39	-----	.89	2.53	-----	-----	.13
H	New	0.29	-----	3.81	0.11	-----	0.16	-----	0.47	0.21	-----	-----	-----	-----	0.10
	Used	.67	4.68	16.68	.74	-----	0.26	-----	0.62	2.37	3.20	3.47	-----	-----	.25
I	New	0.33	-----	0.56	----	-----	0.11	-----	0.58	-----	-----	-----	-----	-----	0.10
	Used	.34	1.18	.85	0.58	-----	-----	.12	0.13	.46	0.16	-----	-----	-----	.11
J	New	0.23	-----	0.29	0.07	-----	-----	0.02	0.29	0.06	-----	-----	-----	0.10	0.07
	Used	.56	-----	.37	.11	-----	0.21	-----	.29	1.11	.31	-----	-----	-----	.08
K	New	0.60	-----	9.80	0.28	-----	-----	-----	2.51	-----	-----	-----	-----	-----	0.24
	Used	1.26	0.39	7.30	.56	-----	-----	0.65	-----	1.86	-----	-----	-----	-----	.37

(1) Zn could be due to wear when present with copper, or as an additive when present alone.

(2) P, S, Ca, Ba probably present as additives.

(3) Limit of detection for sample, when - shown, element is less than this value.

TABLE X. - LFW-1 FRICTION AND WEAR TEST RESULTS (LUBRICANT TEMP. 373 K HERTZIAN PRESSURE 0.55 GPa) (DATA FROM REF. 12)

Lubricant code	Coefficient friction after cycle					Avg coeff of friction	Mean coeff of friction	Avg wear scar width, mm	Mean wear scar width, mm	Weight loss, mg			
	400	800	1200	9000	10 000					Block	Ring	Total	Mean
A New	0.047	0.047	0.059	0.065	0.059	0.055	0.053	1.12	1.07	0.5	1.4	1.9	2.0
A Used	.047	.044	.053	.053	.053	.050		1.02		.4	1.7	2.1	
B New	0.026	0.024	0.024	0.029	0.029	0.026	0.027	1.02	0.91	0.5	1.6	2.1	2.0
B Used	.026	.026	.026	.029	.029	.027		.79		.8	1.1	1.9	
C New	0.032	0.026	0.026	0.021	0.021	0.025	0.024	1.14	1.12	0.7	1.7	2.4	3.3
C Used	.026	.021	.018	.029	.018	.022		1.09		2.7	1.4	4.1	
D New	0.026	0.023	0.021	0.015	0.015	0.020	0.017	1.19	1.12	1.4	1.5	2.9	2.4
D Used	.018	.015	.015	.012	.012	.014		1.04		.2	1.7	1.9	
E New	0.029	0.029	0.026	0.026	0.026	0.027	0.035	1.04	0.95	1.3	0.9	2.2	2.0
E Used	.044	.041	.041	.041	.041	.042		.86		.2	1.6	1.8	
F New	0.044	0.044	0.041	0.056	0.059	0.049	0.034	1.02	1.02	1.4	1.7	3.1	2.5
F Used	.024	.021	.018	.015	.015	.019		1.02		.1	1.7	1.8	
G New	0.047	0.047	0.053	0.071	0.071	0.058	0.043	1.22	1.10	1.1	1.3	2.4	2.0
G Used	.026	.026	.026	.032	.032	.028		.97		.2	1.4	1.6	
H New	0.021	0.021	0.015	0.018	0.021	0.019	0.022	0.99	1.11	1.4	1.8	3.2	2.8
H Used	.029	.026	.024	.024	.021	.025		1.22		.2	2.1	2.3	
I New	0.035	0.029	0.026	0.041	0.041	0.034	0.034	1.19	1.21	1.1	1.3	2.4	2.6
I Used	.038	.038	.038	.029	.029	.034		1.22		.3	2.5	2.8	
J New	0.047	0.047	0.041	0.026	0.026	0.037	0.031	1.07	1.06	0.3	1.0	1.3	1.9
J Used	.029	.029	.029	.018	.018	.025		1.04		1.0	1.5	2.5	
K New	0.024	0.021	0.018	0.015	0.015	0.019	0.022	1.12	1.21	0.4	1.0	1.4	1.5
K Used	.035	.029	.026	.018	.018	.025		1.30		.1	1.4	1.5	

TABLE XI. - MEASURED EFFICIENCIES  
(DATA FROM REF. 1)

Lubricant code	Efficiency	Inlet temp., K
A	0.9840	361.5
	.9850	375.0
B	.9833	356.8
	.9843	375.0
C	.9876	356.4
	.9873	371.5
D	.9860	356.1
	.9874	370.1
E	.9835	361.0
	.9832	371.5
F	.9865	355.7
	.9877	372.0
G	.9873	378.7
H	.9870	355.6
	.9879	372.1
I	.9864	355.6
	.9882	372.2
J	.9864	355.6
	.9877	372.3
K	.9869	355.6
	.9882	372.3

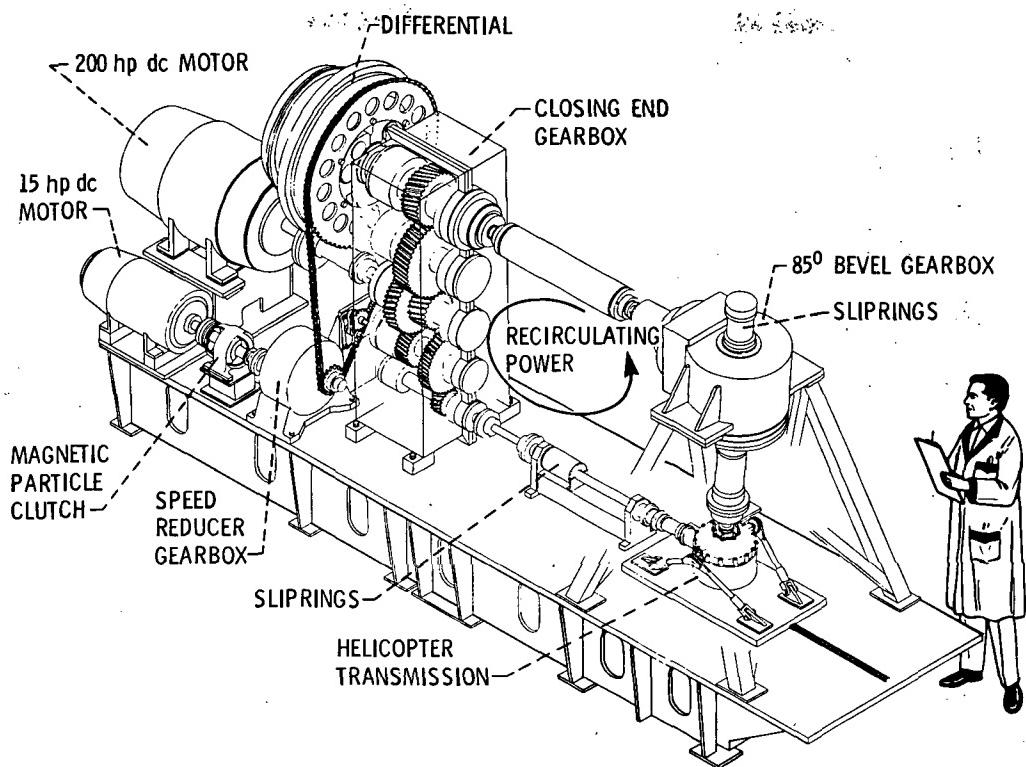


Figure 1. - NASA 500 hp helicopter transmission test stand.

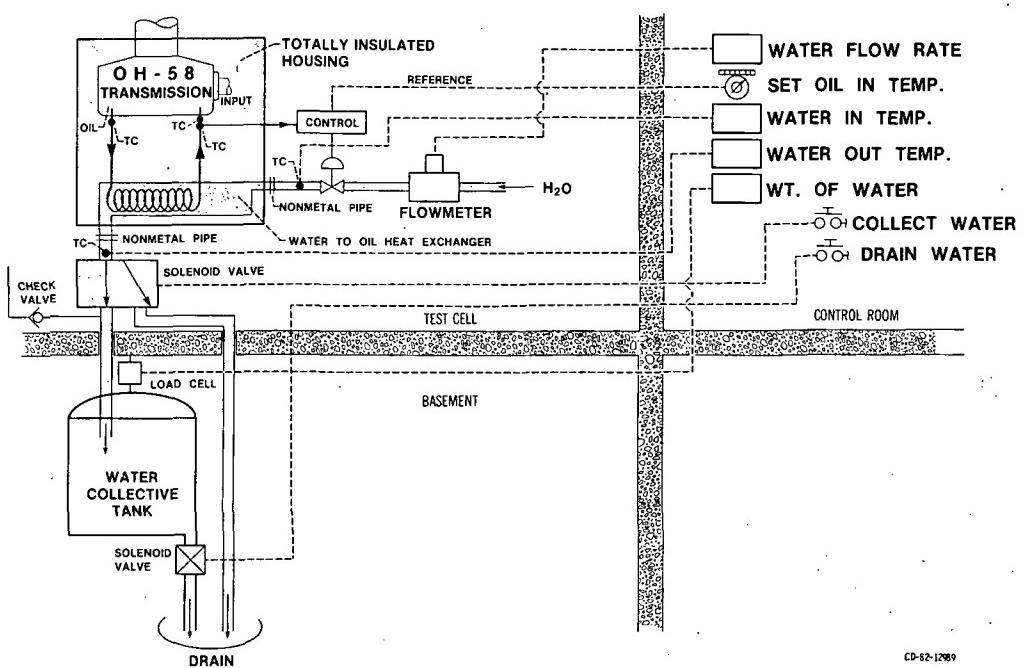


Figure 2. - Schematic of measurement system.

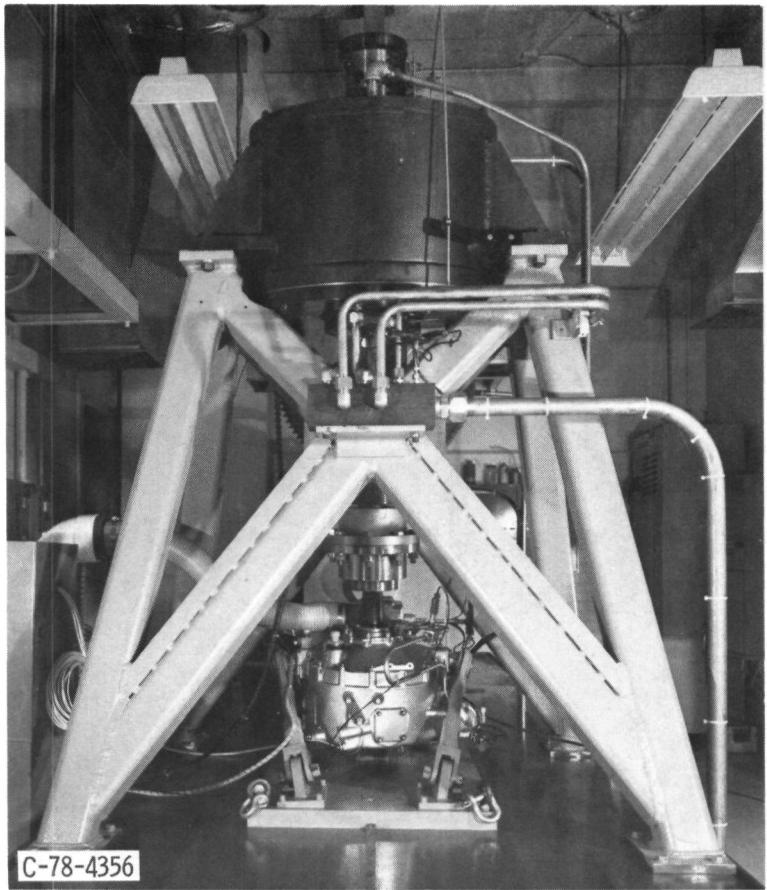


Figure 3. - View of test stand showing OH-58 transmission installed.

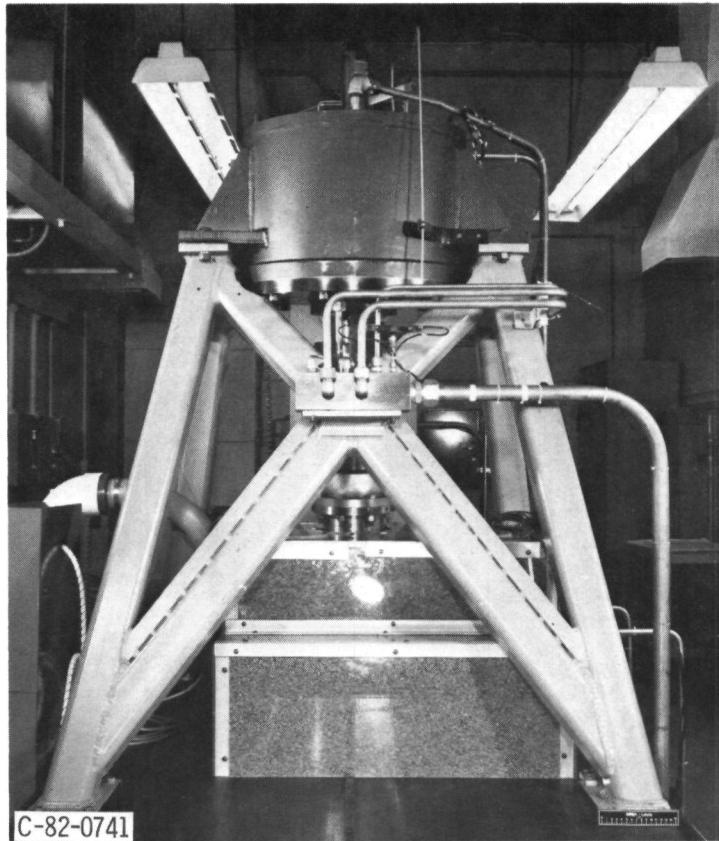


Figure 4. - View of test stand showing insulated transmission housing.

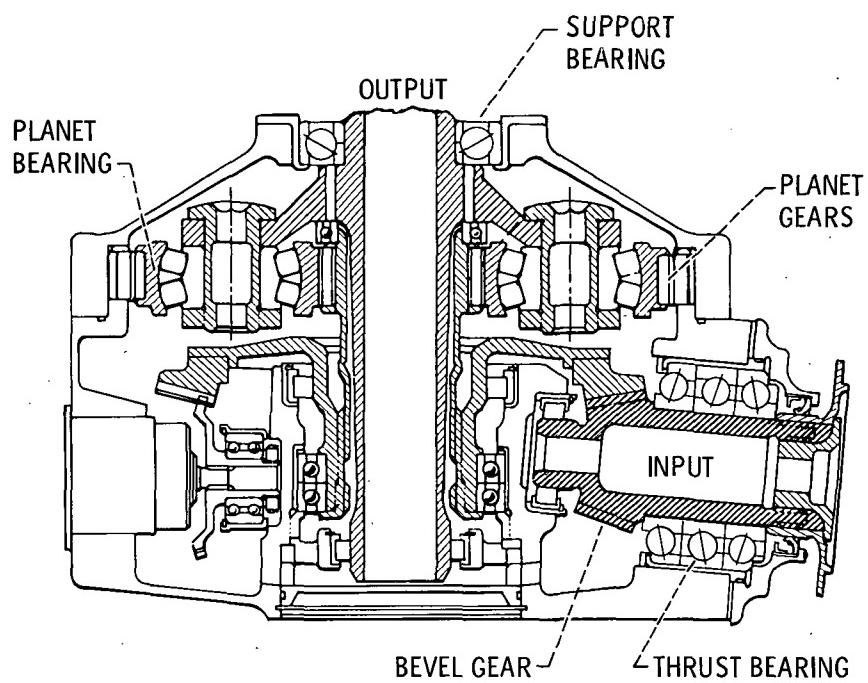


Figure 5. - Cross section of OH-58 helicopter transmission.

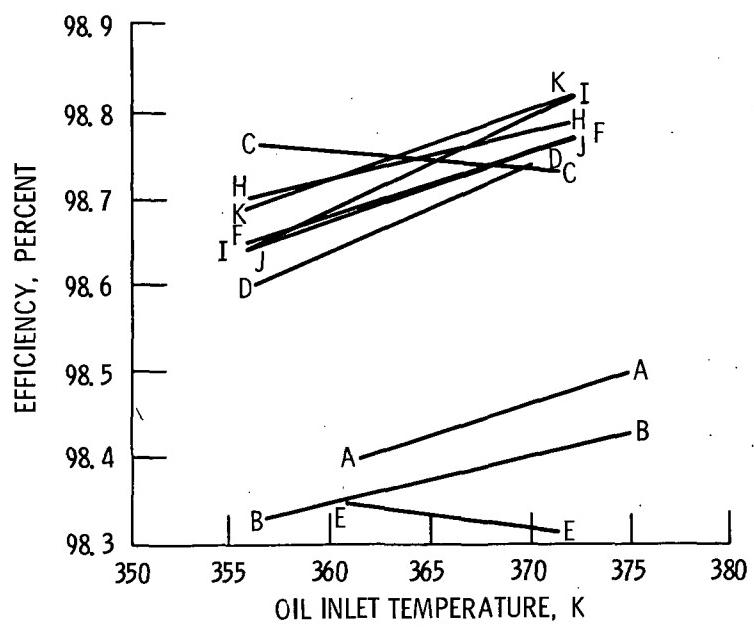


Figure 6. - Experimental efficiency correlated with inlet oil temperature.

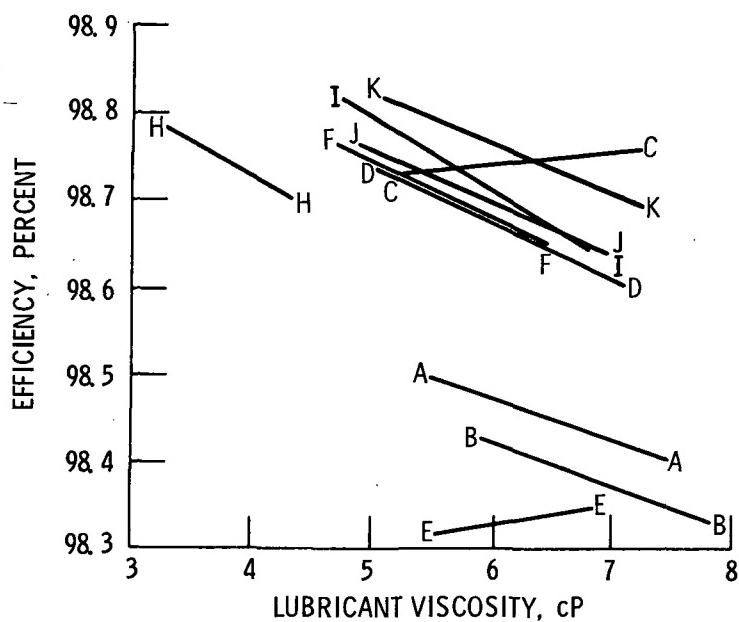


Figure 7. - Experimental efficiency correlated with lubricant viscosity.

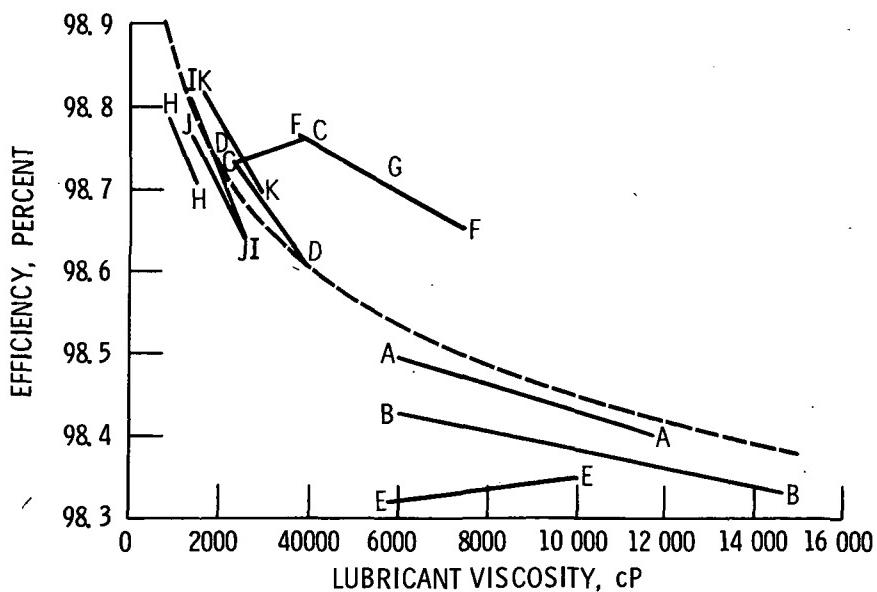


Figure 8. - Experimental efficiency correlated with lubricant viscosity at 0.6 GPa pressure and inlet temperature conditions.

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